# Chapter 2
# Pump Types and Performance Data

**Abstract** Centrifugal pumps are used for transporting liquids by raising a specified volume flow to a specified pressure level. Pump performance at a given rotor speed is described by the rate of flow delivered, the pressure rise achieved, the power absorbed at the coupling, the efficiency and the NPSH. Depending on the application a broad variety of pump types are offered on the market. All of these have at least one impeller and a collector where most of the kinetic energy at the impeller outlet is converted into static pressure. Different forms of impellers, diffusers, volutes and inlet casings are available to build radial, semi-axial or axial, single- or multistage, pumps mounted in horizontal or vertical position – as most suitable for the specific application.

## 2.1 Basic Principles and Components

Centrifugal pumps are turbomachines used for transporting liquids by raising a specified volume flow to a specified pressure level. The energy transfer in turbomachines is invariably based on hydrodynamic processes for which characteristically all pressure and energy differences are proportional to the square of the circumferential rotor speed. By contrast, positive displacement pumps (e.g. piston pumps) essentially deliver the same volume $V_{stroke}$ at each stroke independently of flow velocity or rotor speed n. The flow rate then becomes $Q = n \times V_{stroke}$; the pressure rise results solely from the imposed back pressure.

A centrifugal pump according to Fig. 2.1 is essentially composed of a casing, a bearing housing, the pump shaft and an impeller. The liquid to be pumped flows through the suction nozzle to the impeller. The overhung impeller mounted on the shaft is driven via a coupling by a motor. The impeller transfers the energy necessary to transport the fluid and accelerates it in the circumferential direction. This causes the static pressure to increase in accordance with kinetics, because the fluid flow follows a curved path (Sect. 1.4.1). The fluid exiting the impeller is decelerated in the volute and the following diffuser in order to utilize the greatest possible


J. F. Gülich, *Centrifugal Pumps,*
DOI 10.1007/978-3-642-40114-5_2, 

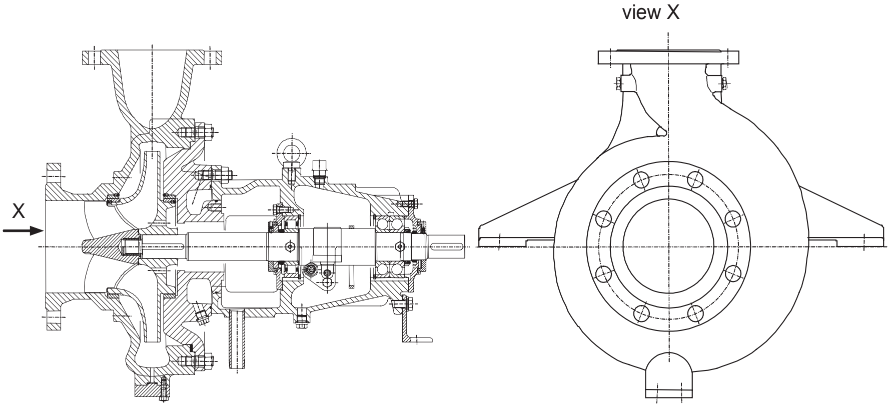



**Fig. 2.1** Single-stage volute pump with bearing frame, Sulzer pumps

part of the kinetic energy at the impeller outlet for increasing the static pressure. The diffuser forms the discharge nozzle.

A shaft seal, e.g. a stuffing box or a mechanical seal, prevents the liquid from escaping into the environment or the bearing housing (the shaft seal is not represented in Fig. 2.1). As shown in Fig. 2.1, an inducer may be added at the impeller inlet for improving the suction performance (Sect. 7.7). However, most applications do not use an inducer.

Impeller and casing are separated by a narrow annular seal through which some leakage flows back from the impeller outlet to the inlet. A second annular seal on the rear shroud serves the purpose of counterbalancing the axial forces acting on the impeller front and rear shrouds. The leakage through this seal flows back into the suction chamber through “axial thrust balance holes” which are drilled into the rear shroud.

The impeller can be described by the hub, the rear shroud, the blades transferring energy to the fluid and the front shroud. In some applications the front shroud is omitted. In this case the impeller is termed “semi-open”.

Figure 2.2 shows the meridional section and the plan view of an impeller. The leading face of the blade of the rotating impeller experiences the highest pressure for a given radius. It is called pressure surface or pressure side. The opposite blade surface with the lower pressure accordingly is the suction surface or suction side. When looking into the impeller eye we see the suction surface. Therefore, it is sometimes called the “visible blade face” or the “lower blade face”, whilst the pressure surface, not visible from the impeller eye, is called the “upper blade face”. These terms are ambiguous and should be avoided. Also defined in Fig. 2.2 are the leading edge LE and the trailing edge TE of the blade. Figure 2.3 shows an impeller designed by means of a 3D-CAD program (the front shroud is removed).

According to the performance and application requirements a wealth of different pump types is available which can be classified by different aspects. Table 2.1 Gives an overview of the different types of impellers, diffusing elements, inlet cas-

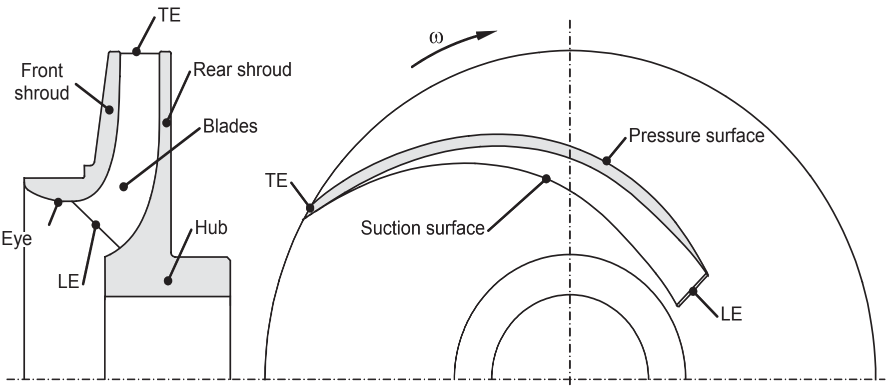
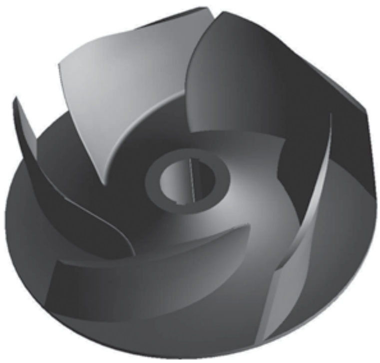



**Fig. 2.2** Meridional section and plan view of a radial impeller, *LE*: Leading edge, *TE*: Trailing edge

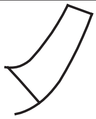
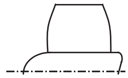
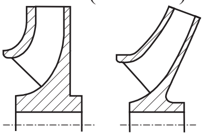
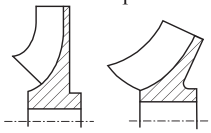
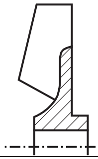
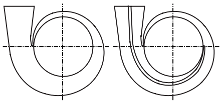
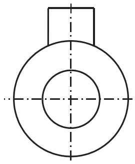
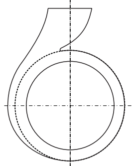
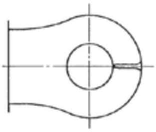
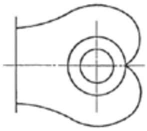
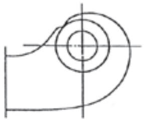
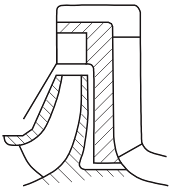
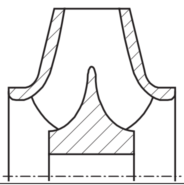
**Fig. 2.3** Radial impeller $n_q = 85$, 3D-model, Sulzer Pumps



ings and combinations of these elements. The following points A to G relate to the corresponding frames in Table 2.1:

A. Depending on the direction of the flow at the impeller exit we distinguish radial, semi-axial and axial impellers. Accordingly the terms radial, semi-axial and axial pumps are used; the latter are also called "propeller pumps".
B. Impellers with a front shroud are called "closed impellers", those without a front shroud are termed "semi-open impellers" and those with large cut-outs in the rear shroud are designated as "open impellers".
C. According to the flow direction at the diffuser inlet there are radial, semi-axial and axial diffusers. Vaneless diffusers are rarely built.
D. The most frequent type of diffusing element for a single-stage pump is a volute casing. Sometimes there is a concentric annulus or a combination of annulus and volute.
E. Most single-stage, single-entry pumps have an axial inlet nozzle as in Fig. 2.1. Inline pumps and between-bearing impellers require radial inlet chambers (Figs. 2.5–2.13). Vertical pumps are often fed via a suction bell (also called "bell-mouth") from a sump in a wet pit installation (see Fig. 2.15 and Sect. 11.7.3).

**Table 2.1 Hydraulic pump components and arrangements**

| Component or feature | Radial | Semi-axial | Axial |
|---|---|---|---|
| (A) Impeller form: Characteristic: flow direction at impeller exit |  |  |  |
| (B) Impeller type | Closed (shrouded)  | Semi-open  | Open  |
| (C) Diffuser Characteristic: flow direction at diffuser inlet | Radial  | Semi-axial  | |
| (D) Outlet casing | Single-volute Double-volute  | Concentric annulus  | Concentric annulus plus volute  |
| (E) Inlet casing | Annular inlet chamber  | Symmetric inlet  | Asymmetric inlet  |
| (F) Impellers in series: multistage pumps |  | Multistage semi-axial pumps see Fig. 2.14. | |
| (G) Impellers in parallel: double-entry impellers | Single-stage  | Multistage double-entry pumps see Fig. 2.13. | |

F. If the pressure generated by one impeller is insufficient, several impellers are arranged in series resulting in a "multistage" radial or semi-axial pump. In that type of pump the diffusers include return vanes, which direct the fluid to the impeller of the subsequent stage. Multistage pumps may be equipped with double volute casings instead of diffusers; in that case the fluid is directed to the subsequent stage through appropriately shaped channels.
G. Double-entry radial impellers are used when greater flow rates must be transported. Double-entry pumps may be built as single-stage or multistage.

The components and characteristics shown in Table 2.1 may be combined in many ways for the optimization of pumps for different requirements. Performance data, design specifications, fabrication methods, installation and operation conditions all have an impact on that optimization.

Because of their complicated, three-dimensionally curved surfaces, impellers, volute casings and diffusers are usually fabricated as castings. In some applications impellers and diffusers are manufactured by NC-milling. Small pumps are often made from plastics. There are even impellers and diffusers made entirely from sheet metal. Although simplified hydraulic shapes may be required to ease their manufacturing, the efficiencies achieved with sheet metal pumps are quite good because of the smooth hydraulic surfaces and the small vane blockage.

## 2.2 Performance Data

The performance data of a centrifugal pump are described by:

- the flow rate Q which is normally defined as the useful volume flow through the discharge nozzle
- the specific work Y or the head $H = Y/g$
- the power consumption P at the pump coupling ("brake horsepower")
- the efficiency $\eta$ at the pump coupling
- the net positive suction head NPSH at the pump inlet, or the net positive suction energy $NPSE = g \times NPSH$.

In addition to these data, the speed n of the pump rotor is indispensable.

### *2.2.1 Specific Work, Head*

The specific work Y is the total *useful* energy transmitted by the pump to the fluid per unit of mass. Y is measured between the suction and the discharge nozzle. Y is identical to the total useful (isentropic) enthalpy rise $\Delta h_{tot}$. In incompressible flow we have $Y = \Delta h_{tot} = \Delta p_{tot}/\rho$ (see Sect. 1.2.2). In practice the head $H = Y/g$ is commonly used (also termed "total dynamic head"). It has to be comprehended as specific *energy unit* (or specific work):

$$Y = \Delta h_{tot} = \frac{p_{2,tot} - p_{1,tot}}{\rho} = g\,H \tag{2.1}$$

The total pressure according to Eq. (1.7) consists of the static (or system) pressure p, the pressure corresponding to the geodetic head $\rho \times g \times z$ and the stagnation pressure $\frac{1}{2}\rho \times c^2$. With reference to Table 2.2, the total dynamic head of a pump measured between the suction and the discharge nozzles results from the difference of the total pressures expressed as heads $H = H_d - H_s$ (subscript d = discharge nozzle; s = suction nozzle).

$$H = \frac{p_d - p_s}{\rho\,g} + z_d - z_s + \frac{c_d{}^2 - c_s{}^2}{2g} \tag{2.2}$$

In this equation all energies are expressed as "energy heads": the static pressure heads $p/(g \times \rho)$ measurable at the suction or the discharge nozzle, the potential energy z and velocity heads $c^2/(2g)$. Head and specific work are *independent* of the density or the type of the medium. Thus a pump (in theory) produces the same head when transporting water, mercury or air. But by no means does it create the same pressure rise $\Delta p = \rho \times g \times H$ that could be measured by a manometer. *All pressure differences, powers, forces and stresses are proportional to the density*.

Table 2.2 shows how the different components that make up the head must be taken into account in a measurement or a calculation. The plane of reference should be chosen at the shaft centerline in the case of horizontal pumps. For pumps with a vertical or inclined shaft, the intersecting point of the shaft axis with a horizontal line drawn through the center of the inlet of the upper suction impeller is used as plane of reference, [N.1]. Manometer readings at other levels should be corrected to the plane of reference, [N.2]. As only pressure *differences* are taken into account for calculating the head, values of absolute pressure or gage pressure (i.e. pressure above atmospheric) may be used in Eqs. (T2.2.1 to 2.2.6).

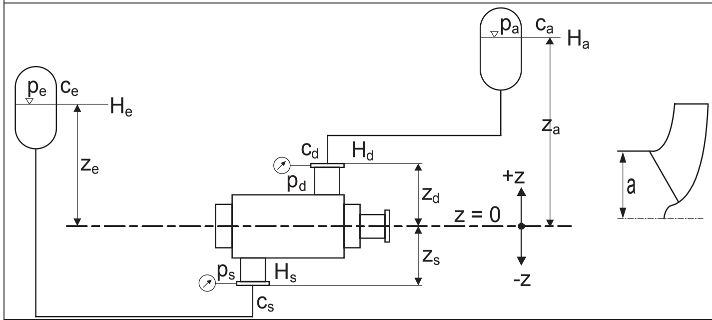
In order to ensure the specified volumetric flow rate through a given pumping plant, the pump must deliver a certain head which is called the required head $H_A$ of the plant. It is calculated from Bernoulli's equation taking into account all head losses in the system (except losses in the pump), see Table 2.2, Eq. (T2.2.6). During steady operation the head of the pump equals the required head of the plant: $H = H_A$.

### 2.2.2 Net Positive Suction Head, NPSH

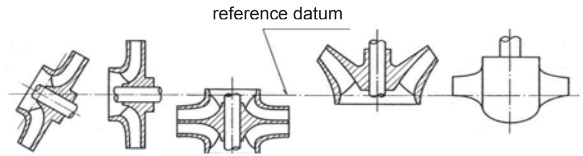
When the pressure in a liquid drops below the vapor pressure, a portion of the fluid will evaporate. Excess velocities due to the flow around the blade leading

**Table 2.2 Total dynamic head and net positive suction head (NPSH)**

| | Pump | Eq. | Plant | Eq. |
|---|---|---|---|---|
| Head at inlet | $H_s = \frac{p_s}{\rho g} + z_s + \frac{c_s^2}{2g} = H_e - H_{v,s}$ | 2.2.1 | $H_e = \frac{p_e}{\rho g} + z_e + \frac{c_e^2}{2g}$ | 2.2.2 |
| Head at outlet | $H_d = \frac{p_d}{\rho g} + z_d + \frac{c_d^2}{2g} = H_a + H_{v,d}$ | 2.2.3 | $H_a = \frac{p_a}{\rho g} + z_a + \frac{c_a^2}{2g}$ | 2.2.4 |
| Total dynamic head | $H_{tot} = H_d - H_s$ | | $H = H_d - H_s = H_a - H_e + H_{v,d} + H_{v,s}$ | |
| | $H_{tot} = \frac{p_d - p_s}{\rho\, g} + z_d - z_s + \frac{c_d^2 - c_s^2}{2g}$ | 2.2.5 | $H_A = \frac{p_a - p_e}{\rho\, g} + z_a - z_e + \frac{c_a^2 - c_e^2}{2\, g} + H_{v,d} + H_{v,s}$ | 2.2.6 |
| Total dynamic head above vapor pressure $p_v$ | $NPSH = H_s + (p_{amb} - p_v)/(\rho \times g)$ | | $NPSH_A = H_s + (p_{amb} - p_v)/(\rho \times g)$ | |
| with $p_{abs} = p_{amb} + p_s$ | $NPSH = \frac{p_{s,abs} - p_v}{\rho\, g} + z_s + \frac{c_s^2}{2g}$ | 2.2.7 | $NPSH_A = \frac{p_{e,abs} - p_v}{\rho\, g} + z_e + \frac{c_e^2}{2g} - H_{v,s}$ | 2.2.8 |
| Static pressure in suction nozzle | | | $p_{s,abs} + p_{e,abs} + \rho\, g\left(z_e - z_S - H_{v,s}\right) + \frac{\rho}{2}\left(c_e^2 - c_s^2\right)$ | 2.2.9 |
| Maximum allowable geodetic suction head ($z_e$ negative)<br>Minimum required geodetic suction head ($z_e$ positive) | | | $z_e = NPSH_R + H_{v,s} - \frac{p_{e,abs} - p_v}{\rho\, g} - \frac{c_e^2}{2g} + a$ | 2.2.10 |




- Note sign (positive or negative ) of all levels z
- $H_{v,s}$, $H_{v,d}$ head losses in suction and discharge pipes respectively
- $p_s$, $p_d$ static pressures measured at suction and discharge branch resp.
- $p_e$, $p_a$ static pressures above suction and discharge liquid levels, resp.



edge cause a local pressure drop, which may lead to such partial evaporation. This phenomenon is called "cavitation"; it is discussed in detail in Chap. 6. Extensive cavitation can impair the performance or even interrupt the flow. Therefore the approach flow conditions at the suction nozzle are an important criterion for the layout and selection of a pump. The relevant parameter is the "net positive suction energy" NPSE or the "net positive suction head" NPSH. It is defined as the absolute suction head $H_{s,abs}$ minus the vapor pressure expressed as head $p_v/(\rho \times g)$. Table 2.2 gives the equations and definitions. We distinguish between the (usually measured) *NPSH of the pump* which is necessary in order to fully or partially suppress cavitation ("NPSH required" or $NPSH_R$) and the *NPSH available in the plant* ($NPSH_A$). Since the vapor pressure $p_v$ is given in the water/steam tables as an absolute pressure, absolute pressures must be inserted into Eqs. (T2.2.7 and 8) for calculating the NPSH.

From Bernoulli's equation we can calculate the absolute pressure at the highest point of the impeller situated at a distance "a" above the rotor axis. This pressure must never be allowed to fall below a level at which an unacceptably large volume of vapor would form at the impeller inlet due to cavitation. Any given pump has its required $NPSH_R = f(Q)$ which corresponds to a *specific amount of cavitation*. The $NPSH_A$ or the liquid level $z_e$ which is necessary for the plant to operate properly must be calculated from the condition $NPSH_A > NPSH_R$, Eq. (T2.2.10), (Chap. 6).

- If $z_e$ calculated from Eq. (T2.2.10) results as *negative*, this value is the *maximum admissible geodetic suction lift:* $|z_{s,geo,max}| = -z_e$.
- If Eq. (T2.2.10) yields a *positive* value, the pump needs a geodetic suction head, which means that the liquid level must be *above* the pump. As demonstrated by Eq. (T2.2.10), this always applies to the pumping of saturated (boiling) liquids, because in that case $p_{e,abs} = p_v$.

With the exception of some special pump types, centrifugal pumps must be filled with liquid for start-up; they are not "self-priming" and thus cannot evacuate the air from the suction pipe.

Types of self-priming pumps include side channel and liquid-ring pumps. Radial impellers are sometimes combined with one side-channel stage in order to allow self-priming of radial pumps, Sect. 2.3.4.

### 2.2.3 Power and Efficiency

Since the specific work represents the energy transferred per unit mass, the *useful power* $P_u$ of a pump is obtained by multiplying the transported mass flow $m = \rho \times Q$ by the specific work Y:

$$P_u = \rho\, Y\, Q = \rho\, g\, H\, Q = Q\, \Delta p \tag{2.3}$$

The power P needed at the coupling is greater than the useful power because it includes all losses of the pump. The ratio of both values is the pump's efficiency η:

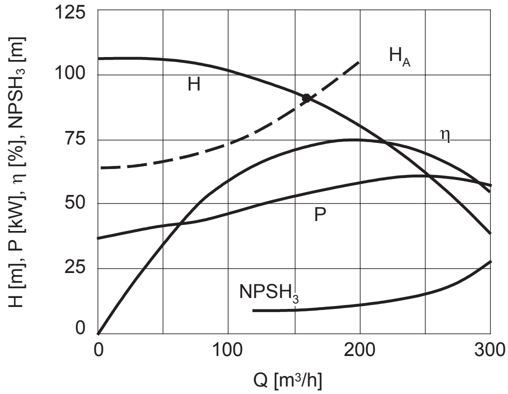



**Fig. 2.4** Pump characteristics and system characteristic $H_A$ (broken curve)

$$\eta = \frac{P_u}{P} = \frac{\rho\, g\, H\, Q}{P} \tag{2.4}$$

### *2.2.4 Pump Characteristics*

When the flow rate of a pump varies, the head, the power consumption and the efficiency change too. Plotting these quantities against the flow rate we obtain the "pump characteristics", Fig. 2.4. At a certain flow rate the pump efficiency has a maximum value called the "best efficiency point" (BEP). The pump is designed for this BEP which is characterized by $Q_{opt}$, $H_{opt}$, $P_{opt}$ and $\eta_{opt}$ at a given speed.

The operation point of a pump is invariably where the head generated by the pump equals the head required by the plant: $H = H_A$. In other words it is at the intersection of the pump characteristic with the system characteristic, refer to Fig. 2.4 and for a detailed discussion to Sect. 11.1.

## 2.3 Pump Types and their Applications

### *2.3.1 Overview*

Centrifugal pumps are of eminent technical and economic importance in many areas of life and industry (the world market volume for centrifugal pumps is in the order of 20 billion US$ per year). Their application range comprises small pumps of a few watts, such as central heating circulation pumps or cooling water pumps for automobiles, as well as 60-MW storage pumps and pump turbines with more than 250 MW when operating in the pumping mode.

**Table D2.1 Definitions of specific speeds**

| **1.** Common in Europe, most used throughout this book | **2**. US customary units | **3.** Truly dimensionless representation | Eq. |
|---|---|---|---|
| $n_q = n\dfrac{\sqrt{Q_{opt}/f_q}}{H_{opt}^{0.75}}$ | $N_s = n\dfrac{\sqrt{Q_{opt}/f_q}}{H_{opt}^{0.75}} = 51.6\,n_q$ | $\omega_s = \dfrac{\omega\sqrt{Q_{opt}/f_q}}{\left(g\,H_{opt}\right)^{0.75}} = \dfrac{n_q}{52.9}$ | 2.5 |
| n in rpm<br>Q in $m^3/s$<br>H in m | n in rpm<br>Q in gpm<br>H in ft | $\omega$ in 1/s<br>Q in $m^3/s$<br>H in m | |

Notes: (1) $H_{opt}$ is the *head per stage*: $H_{opt} = H_{tot,opt} / z_{st}$
(2) In the US the specific speed of double-entry pumps is usually calculated with the *total* flow rate; in this case the factor $f_q$ must be omitted from Eq. (2.5).

The term "centrifugal pumps" comprises radial, semi-axial and axial pumps, but also side channel, peripheral and liquid-ring pumps whose working principles are fundamentally different from that of the first group.

Centrifugal pumps in the narrow sense are designed for flow rates from 0.001 to 60 $m^3/s$, heads of 1 to 5,000 m and speeds from a few hundred to about 30,000 revolutions per minute. These values are intended to illustrate the broad range of applications; they do not define the absolute limits of actual or future pumps.

Any pump application is characterized by the flow rate $Q_{opt}$, the head $H_{opt}$ and the rotor speed n. To a large extent these parameters determine which impeller type and pump design to choose. As will be shown in Sect. 3.4, these three performance parameters are interrelated by the "specific speed" $n_q$, $N_s$ or $\omega_s$ which is of great importance for the selection and design of a pump. Equation (2.5) in Table D2.1 defines $n_q$, $N_s$ and $\omega_s$. In Europe $n_q$ is most popular, while $N_s$ is defined in US customary units. The truly non-dimensional quantity $\omega_s$ should be preferred for theoretical considerations or for deriving general equations, but is not yet used in practice by the majority of pump engineers. This is because the available documentation is often prepared with either $n_q$ or $N_s$. Unfortunately, a variety of other definitions of the specific speed can be found in the literature.

$H_{st} = H_{tot}/z_{st}$ is the head per stage where $z_{st}$ is the number of stages. $f_q$ is the number of impeller entries, that is $f_q = 1$ for single-entry and $f_q = 2$ for double-entry impellers. The choice of radial, semi-axial or axial impellers depends on the specific speed as well as on the pump type. A pump with medium specific speed (e.g. $n_q = 60$) may be built with a radial or a semi-axial impeller depending on the pump type which is most economical for the expected application. From Eq. (2.5) it can be seen that:

- For supplying small flow rates at high pressures, pumps with low specific speeds are required. Below $n_q < 20$ the efficiency drops rapidly with diminishing specific speed (Sect. 3.9, Figs. 3.26–3.32). Therefore, the lower economic limit for a centrifugal pump application is generally $n_q = 5$ to 8 for small pumps, but $n_q = 10$

http://www.springer.com/978-3-642-40113-8

Centrifugal Pumps
Gülich, J.F.
2014, XLI, 1116 p. 714 illus., Hardcover
ISBN: 978-3-642-40113-8